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ON THE CAUSES OF DISCONTINUITY IN STAGE CHARACTERISTICS OF AN AXIAL COMPRESSOR WITH A RELATIVELY LARGE HUB DIAMETER

G.A. Borisov, E.A. Lokshtanov
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ON THE CAUSES OF DISCONTINUITY IN STAGE CHARACTERISTICS OF AN AXIAL COMPRESSOR WITH A RELATIVELY LARGE HUB DIAMETER

by G. A. Borisov, E. A. Lokshtanov, L. E. Ol'shteyn.

ABSTRACT: When the hub diameter of an axial compressor is relatively large, the characteristics of axial stages undergo a break in the usual tests (stage - capacity - throttle, intake and ejection of air in the atmosphere). Missing in such cases are the transitional segments of characteristics between the right branch, which corresponds to the continuous flow-around, and the left branch, which reflects pressure characteristics.
The present article shows that this phenomenon is related to the particulars of the formation of a breakaway flow in stages with a relatively large hub diameter. This, however, occurs not directly but only as a result of the effect of flow separation upon the stability of the entire stage - grid system.

BREAK IN THE CHARACTERISTICS--A RESULT OF STATIC INSTABILITY OF THE SYSTEM.

Fig. 1 shows the experimental characteristics of a stage of an axial compressor having a relative hub diameter at the input of $\bar{d}_1=0.875$. /134*
The data were obtained in tests involving the drawing of air from the atmosphere and its ejection, beyond the throttle, back into the atmosphere. The extent of pressure increase in the stage is $\pi^* \approx 1.2$, while the drop at the throttle is much less than critical. Point A in the characteristic curve corresponds to the full opening of the throttle. As the latter is being shut off, the amount of air used declines monotonically down to a rate reflected by point B. Any further turning off of the throttle--no matter how small--will result in a disproportionate further drop in both the amount of air used and in pressure. The operational parameters of the stage thus move to point C. This transition is characterized by the appearance in the stage of a fully-developed, monozonal, rotating break-away flow. The velocity of relative rotation of this break-away zone is $\bar{\omega} = 0.13$.

*Numbers in the margin indicate pagination in the foreign text.

If now the cross-section of the throttle is increased, the flow of air through the stage will monotonically increase to point D. Next, the flow rate will jump to point E, where the characteristic curve of the throttle, passing through point D, intersects the right, uninterrupted branch of the pressure characteristic of the stage. Stable work of the compressor in the range of operating parameters between points B and D is impossible. /135

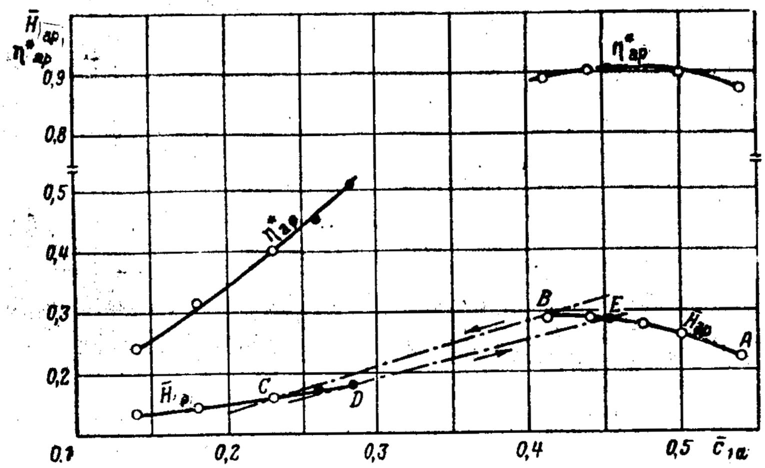

Fig. 1. Characteristic of stage No. 1 obtained in tests following the usual pattern.

—·—·— Characteristic of the throttle in a pre-critical drop of throttle pressure.

It is natural to assume that this "break" in the characteristic is caused by static instability (Ref. 2) of the working parameters of the stage in the given area, i.e. that the pressure characteristic in this sector of the stage is steeper than that of the throttle:

$$F' > \Phi',$$

where $F' = \frac{\partial \pi^*/\pi^*}{\partial G_B/G_B}$ is the slope of the pressure curve of the stage, while

$\Phi' = \frac{\partial p^*_{thr}/p^*_{thr}}{\partial G_{thr}/G_{thr}}$ is the slope of the throttle curve.

To verify this assumption and to obtain the missing segments of the pressure curves which would correspond to the "breaks" in usual trials, special tests were carried out with three stages having a $\bar{d}_1$ from 0.75 to 0.875, with air input direct from the atmosphere but the ejection beyond the throttle. The parameters of the test stages are shown in table 1.

A reduction in pressure beyond the throttle was obtained by means of air suction which made it possible to increase significantly the degree of throttle turn-off and thus increase the steepness of the corresponding characteristic curve.

Table 1. /136

Stage	$\bar{d}_1$	$\bar{c}_{1a}$	$\bar{H}_T$	τ	$z_{d.m.}$
№ 1	0,875	0,51	0,27	0,65	2
№ 2	0,750	0,50	0,32	0,50	2
№ 3	0,750	0,50	0,50	0,68	2

These tests confirmed the assumption stated earlier, namely that in all three stages it was in fact possible to obtain all the characteristic points between B and D (Fig. 2). This proves that <u>the break usually found in the characteristic curve of a stage with a relatively large hub diameter is due not to any inherent impossibility to stabilize the air flow through the stage for operating parameters at which the break occurs, but is due to a static instability of these parameters resulting from the great steepness of the stage curve-- i.e. the acute decline of air pressure in the left branch.</u>

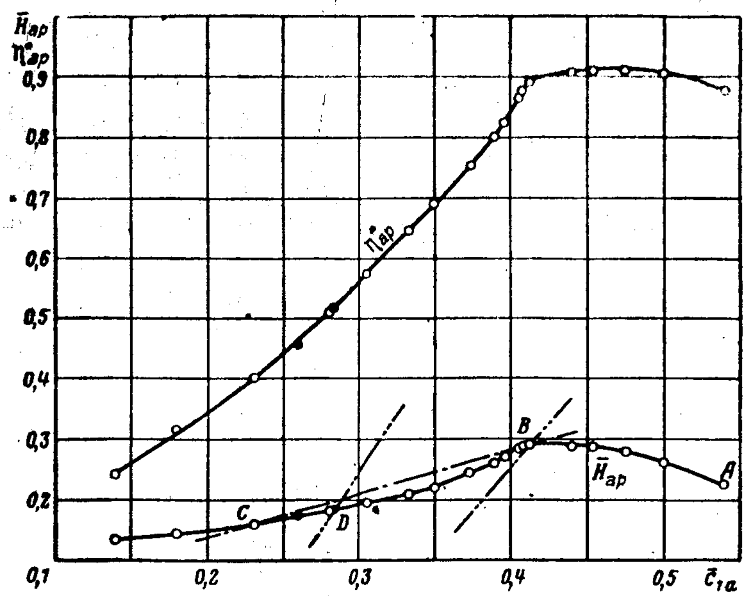

Fig. 2. Characteristic curve of stage No. 1, obtained in tests with air withdrawal beyond the throttle.

—·—·—·—Throttle curve during the pre-critical period of pressure drop in the throttle;

—··—··—Throttle drop during the critical period of pressure drop in the throttle.

However, as was shown in ref. 2, an increase in the steepness of the throttle curve reflects a drop in energy scattered during vibrations, i.e. a deterioration of dynamic stability. In tests with air suction beyond the throttle, the condition of dynamic stability was not satisfied at any point to the left of point B. It turned out that /137

$$F' > L/C\Phi'$$

where L is the equivalent inertia of the system, and
C--its equivalent capacity.
As a result, flutter was generated in the system.

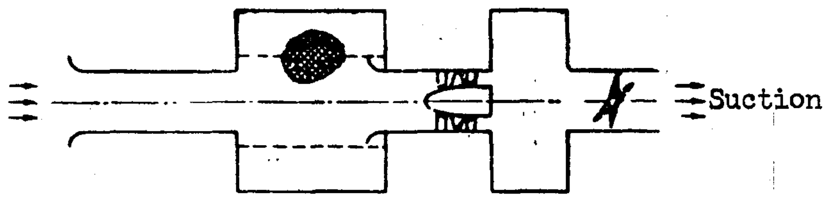

Fig. 3. Diagram of the stage tests. Air is removed by suction from behind the throttle and a damping device operates at the input.

To raise the dynamic stability of the system in this case, the method used was that described in ref. 2. The receiver positioned before the stage was divided into two volumes by lengthwise, concentric nets having a high degree of hydraulic resistance (Fig. 3). Such an arrangement leaves virtually unaltered the characteristics of the net at established operating parameters, but it sharply increases the dissipation of energy when pressure-generated oscillations arise in the receiver. As a result, in tests of stage No. 1, with air suction beyond the throttle and lengthwise nets in the input receiver, it was possible to eliminate flutter over the entire range of operating parameters represented in the characteristic curves. Moreover, at rates which ordinarily correspond to the area of the "break" in the curve, oscillograms were now obtained which make it possible to follow through the progressive development of the break in the blade rims of the stage.

PARTICULARS OF THE DEVELOPMENT OF A STAGE BREAK WHEN THE HUB DIAMETER, $\bar{d}_1$, IS RELATIVELY LARGE.

As the amount of air passing through a stage declines with the development of a "break" in the blade rims, the pressure in the stage drops. The extent of this drop varies in stages with different parameters.

It is a known fact that the diameter $\bar{d}_1$ and the coefficient of air expenditure $\bar{c}_{1\alpha}$ are the two factors having the greatest effect upon the development of the stage characteristic curve to the left of point π^*_{max}.

In stages with a small hub diameter $\bar{d}_1$, parameters such as pressure, work performed, coefficient of pressure drop, all change smoothly with a decline of air expenditure in the area of the break, without any abrupt changes or discontinuities. (Fig. 4). Pressure measurements with the aid of low inertia equipment as well as flow- /138
temperature measurements made in front of the working wheel, show that this smoothness in the case of relatively small hub diameters ($\bar{d}_1 \leq 0.5$) is due to the gradual spread of the break from the periphery of the stage to its hub sections and to a progressive widening of the break-zone along the periphery. The number of such zones also changes, reaching 4 to 5.

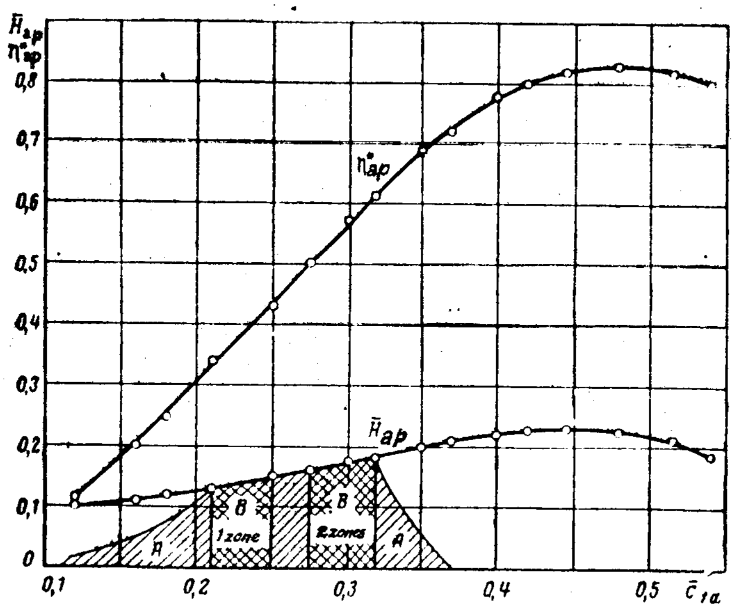

Fig. 4. Characteristic curve of a stage
with $\bar{d}_1$ = 0.4
A--irregular break; B--rotary break.

A rotary monozonal break occurs in stages with a relatively large hub diameter, in the immediate vicinity of the π^*_{max} operating regime. With progressive throttling, the rotation speed of the break-away zone is somewhat reduced, but its peripheral expansion is increased. The breakway zone covers practically the entire height of the blade. Flow pulsation oscillograms--both in front and behind of stage No. 1--as they appear at various operating regimes in the left branch of the characteristic curve, are shown on Fig. 5.

It is precisely these peculiarities of the breakaway flow--its expansion to the full height of the blade and its monozonal structure--that result in a sharper pressure drop when the break occurs in stages with a relatively large hub diameter. Any change in effective pressure following the formation of breakaway zones, can be attributed to two causes: an increase in losses due to the leveling out of the breakaway process behind the stage, and changes in the amount of work expended to increase the pressure of the gas passing through the stage as a result of the spread of the flow near the breakaway zones. The effect /139
of these factors is all the greater, the greater is the area of an individual breakaway zone.

FIGURE NOT REPRODUCIBLE

Fig. 5 (a, b). Pressure pulsation oscillograms of stage No. 1 during the formation of breakaway zones obtained in suction tests as follows:

a) $\bar{c}_{1\alpha} = 0.39$; b) $\bar{c}_{1\alpha} = 0.35$.

1--Full pressure pulsations behind the stage; 2--Full pressure pulsations ahead of the working wheel; 3--Rotation count (14.14 revolutions); 4--Time scale:(0.002 sec.)

The spreading of the flow to the sides of the breakaway zone results in an increase of the angle of attack of the working-wheel blades in the area adjacent to the breakaway zone, on the side opposite to the direction of rotation. In the middle portion of the zone of a break-free flow, the direction of the flow is not affected. (Fig. 6). It can be assumed that, as the amount of air used in the stage is reduced, the blades carrying the greatest load, located near the break- /140
away zone on the side opposite to the direction of rotation, work under the conditions of a limit load, i.e. with a maximum deviation of the flow which corresponds to $\beta_{1\ min} = \beta_{1c}$.

FIGURE NOT REPRODUCIBLE

Fig. 5 (c, d). Oscillograms of pressure pulsations in stage No. 1 during the formation of a breakaway zone, the pulsations being obtained in suction tests as follows:

c) $\bar{c}_{1\alpha} = 0.28$; d) $\bar{c}_{1\alpha} = 0.23$

Figures 1 to 4 designate the same oscillograms as those appearing in Fig. 5 (a, b).

If, in a system of coordinates rotating jointly with a breakaway zone, the flow--as a result of spreading--deviates in the vicinity of the zone by the value of angle $\delta\gamma_1$, then, when $\beta_1 = \beta_{1c}$, the coefficient of air expenditure in the no-break zone $\bar{c}_{1ac}$ must be greater than the coefficient of expenditure at the border of the stage break $\bar{c}_{1ac}$ by the value

$$\frac{\delta\bar{c}_{1a}}{\bar{c}_{1ac}} = \frac{\bar{c}'_{1ac} - \bar{c}_{1ac}}{\bar{c}_{1ac}} = \frac{\bar{c}_{1ac}}{\bar{r}_1} \cdot \frac{\delta\gamma_1}{\sin^2\gamma_1} \cdot \frac{1}{1-\bar{\omega}}.$$ /141

The theoretical pressure in the respective sectors of the no-break zone increases proportionately to the rise in $\bar{c}_{1a}$, while on the opposite side of the breakaway zone the pressure will drop by the same amount as the result of an increase of the warping of the flow due to rotation. Even if one assumes that the axial component of velocity in the no-break zone along the periphery is not altered, the mean theoretical pressure in the stage of a no-break flow will still drop, as compared to the pressure on the border of the break, by the value of

$$\delta\bar{H}_{\tau} = \bar{H}_{\tau} - \bar{H}_{\tau.c} = -(r_1^2 - \bar{H}_{\tau.c})\frac{\delta\bar{c}_{1a}}{\bar{c}_{1ac}}.$$

Actually, the drop in the amount of work expended will be even greater since the mean velocity of the no-break zone is always greater near the breaking zone on the side opposite to that of the direction of rotation.

In Ref. 1 it was shown that a reduction in the rotation velocity of the breakaway zone with an increase in the relative hub diameter is due to the increased influence of the joint flow sectors both ahead and behind the stage in question, i.e. to an increase in their relative length.

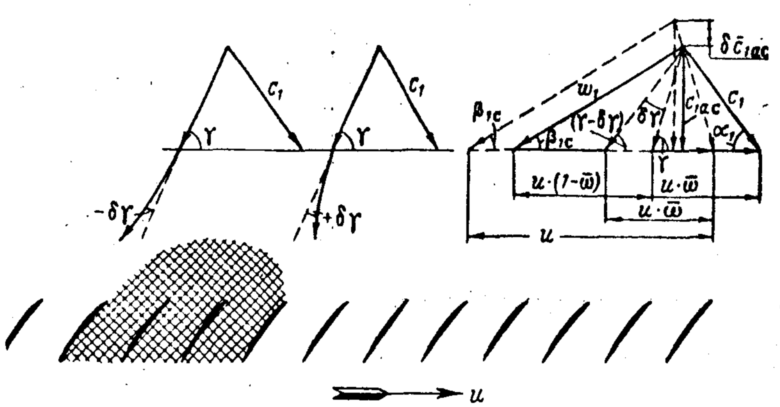

Fig. 6. Diagram of a breakaway flow in the stage of an axial compressor.

If the breakaway zone extends to only a portion of the radius of the flow, then the shift in both the separation flow and in the main flow will occur not only radially but also along the circumference. If, however, the breakaway zone takes up the entire height of the blade, then the displacement can occur only across the lateral borders of the zone. And since the relative area of the lateral surfaces is small, the length of the shift--i.e. the length of the adjoining flow sectors ahead of and behind the stage--grows, while the rotation velocity of the breakaway /142
zone declines. Thus the small rotation velocity of the breakaway zone, as shown in experiments, indicates, in turn, that the breakaway zone in a stage with a large $\bar{d}_1$ extends over the entire height of the blade or, at least, over the greater part of this height.

Let us now consider how to explain the fact that in stages with a relatively large hub diameter the breakaway zone spreads over the entire height of the blade or at least over its greatest part.

We know that, in the working wheel of an axial compressor, retarded particles of the border layer are rejected by centrifugal forces toward the periphery of the blade. For the same reason, low energy particles are also rejected toward the periphery when the border layer breaks away--and regardless of where it had first originated. As a result, the breakaway zone in the working wheel attaches itself to the periphery of the flow zone (fig. 7).

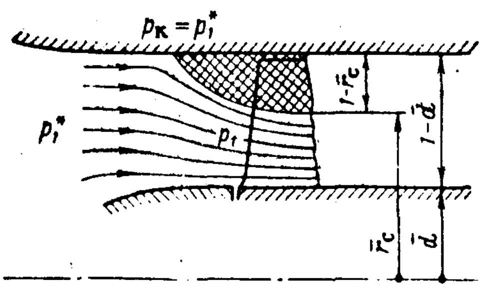

Fig. 7. Diagram of the flow in the breakaway zone.

The mean tangential velocity of the particles of air in the area of the breakaway zone--both at the working wheel and ahead of it--is close to the circular peripheral velocity of the wheel itself. Accordingly, the radial pressure gradient in the zone is determined by centrifugal forces as follows:

$$p_к - p_{r_c} = \int_{r_c}^{R_к} \rho\Omega^2 r dr = \frac{1}{2}\rho\Omega^2 R_к^2 (1 - \bar{r}_c^2). \qquad (1)$$

Here, index "k" refers to the periphery of the flow section, while r_c is the inner radius of the breakaway zone.

At the same time, the pressure along the periphery of the zone must differ but little from the full pressure of the incoming flow in front of the zone, i.e.

$$p_{\kappa} \approx p_1^*, \tag{2}$$

while pressure p_{r_c} on the border of the breakaway zone on the side of the hub coincides with the static pressure in the no-break flow of the same radius:

$$p_{r_c} \approx p_1^* - \rho \frac{c_{1c}^2}{2} \tag{3}$$

By using expressions (1), (2) and (3) to define p_{r_c} and p_{κ}, it is possible to establish a connection between the radius of the inner limit of the breakaway zone and the velocity of the break-free flow: $1 - \bar{r}_c^2 \approx$ $\approx \bar{c}_{1c}^2$, or

$$\bar{r}_c \approx 1 - \frac{\bar{c}_{1c}^2}{2}. \tag{4}$$ /143

Thus the radius of the border of the breakaway zone depends only upon the velocity of the main flow, while the relative portion of the blade height taken up by the break,

$$\frac{1-\bar{r}_c}{1-\bar{d}_1} \approx \frac{\bar{c}_{1c}^2}{2(1-\bar{d}_1)} = \frac{\bar{c}_{1ac}^2}{2\sin^2\alpha_1 \cdot (1-\bar{d}_1)} \tag{5}$$

depends also upon the relative diameter of the stage hub.

This result is in qualitative agreement with numerous experimental findings, for it is known that the extent of pressure drop in the left branch of the characteristic curve is all the greater, the greater are the values of $\bar{d}_1$ and $\bar{c}_{1ac}$. It is difficult to expect a good quantitative correspondence between the results of the above experiment and the formula derived on the basis of purely qualitative considerations. However, a numerical evaluation, too, gives very likely results: if one assumes that the velocity of the flow in a no-break zone is close to the mean velocity just ahead of the working wheel, at the border of a stable, axially symmetrical flow, then it turns out that, for stage No. 1, $(1 - \bar{r}_c) / (1 - d_1) \approx 1$, i.e. the breakaway zone takes up the full height of the blade.

It may be assumed that the portion of the blade height taken up by the breakaway zone increases because of the influence of the directing mechanism. Just as particles of air in the breakaway zone of the border layer in a working wheel are rejected toward the periphery, so do retarded particles in the directing mechanism shift toward the hub.

The low velocity zone near the hub of the directing mechanism raises the counterpressure behind the working wheel and thereby assists the spread of the breakaway zone over a greater portion of the blade height in the working wheel. For this reason it is advisable to introduce, in formula (5), a coefficient K, thus:

$$\frac{\bar{l}_c}{1-\bar{d}_1} = \frac{1-\bar{r}_c}{1-\bar{d}_1} = K \frac{\bar{c}_{1c}^2}{2(1-\bar{d}_1)} . \qquad (6)$$

The value of K must be comprised within the limits of $1 < K < 2$, and the maximum value $K = 2$ corresponds to the assumption that the relative extent of the breakaway zone in the directing mechanism, $\bar{l}_{b.d.m.}$, equals $\bar{l}_{b.w.w.}$ computed as per formula (5), and that the total extent of the breakaway zone along radius $\bar{l}_c = \bar{l}_{b.w.w.} + \bar{l}_{b.d.m.}$

The ratio $\bar{c}_{1ac}^2/[\sin^2\alpha_1 \cdot (1-\bar{d}_1)]$ can be regarded as a criterial parameter which could be usefully employed for an analysis and generalization of experimental data dealing with a change in stage pressure in the left branch of the characteristic curve. /144

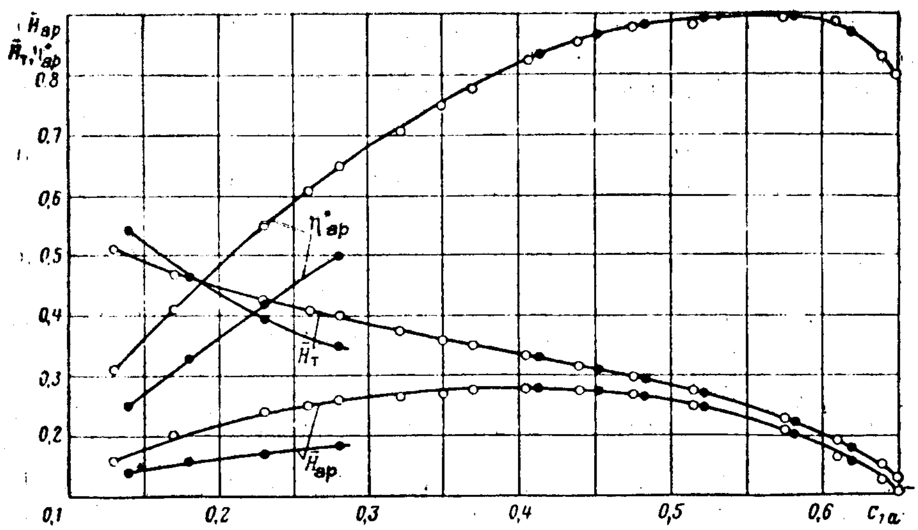

Fig. 8. Characteristic curves of two rims (that of the directing mechanism, D.M., and the working wheel, W.W.) of stage No. 1, obtained in tests by the usual method.

o -- in tests without D.M. • -- in tests with D.M.

From formula (6) it is possible to establish a connection between the minimum value of hub diameter $\bar{d}_1$ at which the breakaway zone takes up the entire height of the blade, and $\bar{c}_{1c}$:

$$\bar{d}_{1\text{кр}}=1-\frac{K}{2}\,\bar{c}_{1c}^{2}. \qquad (7)$$

A special experiment was set up to clarify the role of the directing mechanism in the development of the break and the progress of pressure characteristics of stage No. 1. These characteristics were recorded for the first two rims--those of the directing mechanism and the working wheel--both in the presence of the directing mechanism and without it. The respective curves are shown in Fig. 8. When a directing mechanism is present, the curves of the first two rims of the left branch --as well as those of the stage as a whole--show a break, reflecting a sharp drop in pressure as well as in the coefficient of efficiency. After the removal of the directing mechanism, the left branch of the curve changes sharply--both in terms of pressure and efficiency. It varies smoothly and continuously as air expenditure declines and, for any given rate of air expenditure, both efficiency and pressure register higher values than they do in the presence of a directing mechanism.

Thus the removal of the directing mechanism has led to a qualitative change in the structure of the breakaway flow. Instead of a stable, unizonal break in the stage, a multizonal break is formed with a much greater /145
rotation speed of the breakaway zones (Fig. 9), the number of zones reaches 12, and the relative velocity of their rotation increases from 0.13 to 0.23.

FIGURE NOT REPRODUCIBLE

Fig. 9. Oscillograms of pressure pulsations in stage No.1 obtained in tests without a directing mechanism ($\bar{c}_{1a}$= 0.242): 1--Pulsations at full pressure behind the working wheel. 2--Pulsations at full pressure in front of the working wheel. 3--Time scale (0.002 sec.).

The question of the influence of the directing mechanism and of the number of fixed and rotating rims, in general, upon the rotation speed of breakaway zones was examined earlier in ref. 1. In the present case, in order to understand the causes of pressure drop in the stage following the formation of a break, it is the influence of the directing mechanism upon the structure of the break flow that is of the greatest importance. Two reasons can be cited for this: a) the rise in counter-pressure at the hub, behind the wheel, and b) a drop in the rotation velocity of the breakaway zones.

After the break has spread over the entire height of the blade, the difference of pressure in the breakaway zone, $p_{\text{к}}-p_{\text{вт}}$, determined by centrifugal forces, can be compensated only by a curvature of the flow surfaces in the meridian plane. The setting up of a directing mechanism, involving increased losses at the hub, leads to an increased curvature of the flow lines as well as to increased pressure in the hub sections of the working wheel.

A reduction in the rotation pressure of breakaway zones leads to a situation wherein the flow ahead of the breakaway zone has time to slow down at a considerable distance from the wheel. In this way, the relative length of the sector in which the spreading of the flow occurs near the breakaway zone, increases, and the widening of the relative surface being occupied by a single breakaway zone becomes possible. This, in turn, leads to a greater change of direction in sectors of the /146
main flow adjacent to the breakaway zone and also to a drop in the amount of work being expended to increase flow pressure, i.e. to a still greater drop in stage pressure in the left branch of the characteristic curve.

From experiments with multistage compressors, it is known that the rotation speed of breakaway zones and the structure of a breakaway flow in a given stage can be altered substantially by the presence of other blade rims. In considering this fact in conjunction with the described mechanism of the influence of the breakaway structure upon the changes in pressure, it may be assumed that the form of the left branch of the characteristic curve of the stage also is determined not only by the parameters of the stage itself but also by the influence of adjacent rims and the conditions on the stage boundary in general.

Conclusions

1. A break in the characteristics of a stage of an axial compressor having a large relative hub diameter is caused, in the usual tests, by the static instability of the stage-throttle system ($F' > \Phi'$). The break may be eliminated without altering the stage characteristics by means of increasing the pressure drop at the throttle, i.e. by increasing the steepness of its characteristic curve.

2. The static instability of operating parameters of stages with a large hub diameter $\bar{d}_1$ is due to the acute drop in useful pressure in a

stage following the formation of a breakaway flow. This, in turn, can be explained by the fact that, at large values of the hub diameter, the breakaway zone quickly spreads over the entire height of the flow section of the stage. For a given value of the hub diameter, the portion of blade height taken up by the break is directly proportional to the dimensionless absolute velocity of the flow in front of the wheel, at the border of the no-break operating zone. In quantitative terms, this can be expressed by the following formula:

$$\frac{1-\bar{r}_c}{1-\bar{d}_1} = K\frac{\bar{c}_{1c}^2}{2(1-\bar{d}_1)}.$$

3. The presence,behind the working wheel,of a stationary directing mechanism reduced the velocity of rotation in the breakaway zones and leads to a unizonal structure of the break. This, apparently, increases the spread of the flow on both sides of the breakaway zone and this results in a sharper drop of the pressure in the no-break flow.

In the absence of a directing mechanism in the stage, there is formed a multizonal break and the pressure in the left branch changes smoothly.

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